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ABSTRACT

Roll stabilization of ships by activated fins is becoming increasingly common. Model testing of ships and their appendages has historically been a valuable source of information concerning full-scale behavior. The authors argue that model testing of ship stabilization systems should be equally valuable.

This thesis is a first study of the theoretical relationships and the practical hardware necessary for the operation of model stabilizers. The requirements of dynamic similitude based on the Froude number are developed. Practical considerations for the towing of stabilized models are indicated.

Servo methods are used to show the dynamic behavior of a ship among waves, and the mathematical requirements of a stabilizing system are synthesized. The physical components necessary to satisfy the mathematical relationships are suggested. An analog system for testing the components individually and as a system prior to installation in the model is described. The analog system was operated and its effectiveness is shown.

The instrumentation as developed by the authors has several faults, and corrective measures are suggested. Recommendations for refinement of the work and for future study are made.

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I.

INTRODUCTION

A. Historical Interest and General Comments

Ship stabilization as an art dates back to the nineteenth century. Chadwick[1] has listed in tabular form the historical growth of the art. We note that three principal means of achieving stabilization have been employed: gyroscopes, moving liquids, and fins. These are all active means of control. In addition, such passive elements as moving weights, tuned tanks, and the almost universal bilge keel have been employed.

We shall speak of ship stabilization with reference to angular motion about the longitudinal axis of the ship, referred to hereafter as roll. It is true that a ship has five other degrees of freedom. The surge and sidle modes are not of great consequence. Yaw is largely a problem in steering. Heave is important, but the energy involved is large, and at present there appears to be no satisfactory means of controlling it.

This leaves only the pitching mode to be dealt with. Due to the fact that a ship's longitudinal mass moment

of inertia is much greater than the transverse moment of inertia, the energy exchange in pitching is considerably larger than in rolling, even though the angular amplitudes are generally smaller. This requires that a pitch stabilization system be capable of supplying a high level of energy in order to be effective. Historically, the authors are not aware of any successful attempt at such a system. Technically, the problem seems susceptible to solution.

In addition to the fact that the energy levels involved in rolling are smaller than in pitching, it is generally agreed that rolling contributes more to human discomfort and to physical damage. For these reasons, the major effort has been directed toward roll stabilization.

B. Present State of the Science

It has only been within recent years that ship stabilization has become a science, capable of reasonably exact engineering analysis and solution. The rapid post-war emergence of servo theory has had an important and direct bearing on this development. We now have a useful tool for analyzing the dynamic behavior of a ship in roll and for synthesizing a means of stabilizing the roll.

Of the various mechanical methods of roll stabilization, only activated fins enjoy much popularity at present. The gyroscope is incapable of delivering large torques save at the expense of excessive weight, space, and power requirements.

A system of tanks consumes a considerable internal volume in the ship, even though it is effective at all ship speeds and does not add to the propulsive load of the ship. In spite of the fact that they are ineffective at lower ship speeds and may add to the drag of the ship under certain conditions of sea state and speed, activated fins have advantages too important to ignore. They require less weight, space, and internal power than any other scheme, and in certain conditions of sea state and ship speed may reduce the propulsive drag of the hull.[2]

The body of knowledge surrounding stabilization by activated fins is fairly mature. Fins are afloat in a number of passenger liners and in at least two ships of the United States Navy. Two large industrial firms and several subsidiary concerns enjoy a considerable business in the manufacture of activated fins.

C. Model Testing

The use of scale model testing in marine applications has been of tremendous value. Both towed and self-propelled ship models are tested in tanks, providing information concerning the ships's propulsive characteristics not otherwise available. Ship's propellers, rudders, and other appendages are tested on a model scale. The laws relating the dynamic behavior of the prototype to its model are well known and have been widely published in the literature.

The authors believe that, in light of the proven value of model testing in the past, the application of the same principles to ship stabilization systems should prove of equal value in the future. Stabilizers represent a large financial investment, and the shipowner desires to get as much for his money as possible. The shipbuilder would like to insure that the stabilizing system will meet his guarantees and perform in an optimum manner. From the standpoint of cost and efficiency, such a system must not only be capable of performing its assigned task, but it must be the optimum system capable of being designed. The authors believe that tests of stabilizing systems in ship models will be of considerable value in achieving this result.

Stabilizer design in the past has been based largely on experience with, and the performance of previous designs. This has been augmented in recent years by servo theory. Experience is costly and accumulates slowly. In 1945, Allan(3) devised a scheme for the model testing of activated fins. The fins were manually operated. The results, though gratifying, were not identically reproducible. We cannot expect the human being to simulate the dynamic behavior of an electro-mechanical system. Chadwick(4) reports the use of a model tank stabilization system to corroborate his theoretical developments.

The authors have no knowledge of any record of a complete activated fin stabilizer having been built and operated in model size. It is the authors' thesis that such a model is theoretic-

ally and technically realizable. This paper is an initial study of the physical realization of such a model system. The authors have built a servomechanism which, with certain modifications, should be satisfactory for towing tank work. Its use should permit the naval architect to establish any stabilizer configuration and to vary any parameters which he may desire, thereby affording an additional tool for optimization of the solution of the entire problem.

II.

THE ROLL STABILIZATION PROBLEM

A. Stabilization as a Servo Problem

The stabilization of roll in ships is essentially a problem in control. It is most readily discussed using servo terminology. For this reason, it is necessary to assume that the reader is familiar with servo theory.

A ship consists of three common physical elements encountered in servo work: a mass moment of inertia about the axis of roll; a viscous damping; and a potential energy storage element (spring) due to its natural righting moment. We may linearize the problem by assuming that the roll axis remains fixed, and that the damping and righting coefficients remain constant with angular displacement. These assumptions are reasonably valid for angles of roll less than about ten degrees.

Wave motion exerts a disturbing torque on the ship about its axis of roll. In a ship equipped with stabilizers, the fins would exert a torque in opposition to that of the seas. The transfer function of the ship is:

(1) $$T_r = (J s^2 + B s + K) \Theta$$

T_r = Resultant torque acting

J = Moment of inertia about axis of roll

B = Viscous friction coefficient

K = Righting moment (product of displacement and metacentric height)

Θ = Angle of roll

The block diagram for the complete control system may be constructed in two ways. In both cases, we desire that the angular displacement of the ship be as near to zero as possible. We may first consider the system as a positional servo, in which a zero angle is ordered and wave effect enters as a disturbance. We may next consider the wave torque as an input and roll angle as an output. The components of the two systems are the same; the difference is in the mental approach to the problem. In each case we wish to minimize the effect of the seas on the displacement angle. It will be immediately recognized that, accepting the physical parameters of the ship as fixed, it is necessary to close a feedback loop and make its gain as high as possible.

Let us consider the wave disturbance as an input and angular displacement as an output. In order to close the feedback loop it is necessary to measure certain characteristics of the output and use them to operate on the input. Physically this says that the fin action must be governed by the motion of

the ship. We shall choose to measure angular displacement, rate, and acceleration in order to describe the ship's behavior in roll. Other types of signals could be used to close the feedback loop, such as bending moment on the fin shafts, or the pressure distribution of waves against the ship's sides. The roll angle and its derivatives are most commonly used and most easily measured, however.

These signals must be used to operate the fins. They may be combined in the proper amplitudes in a summing amplifier, which in turn serves to drive a servomotor actuating the fins. As will be shown later, smaller feedback loops will exist in addition to the main loop. However, the basic reduced block diagram for the system just described is shown in Figure 1. In this diagram, T_d is the disturbing torque of the waves; θ is the roll angle of the ship; A, C, and D are gain constants operating on $\ddot{\theta}$, $\dot{\theta}$, and θ, respectively; G(s) is a transfer function between the servomotor control signal and the torque exerted by the fins; T_f is the restoring torque which the fins exert on the ship.

The complete transfer function is

$$\frac{\theta}{T_d} = \frac{1}{J s^2 + B s + K + G(s)\,(A s^2 + C s + D)}$$

If the response of the servomotor is rapid, it may be assumed that G(s) is approximately a constant G, in which case the transfer function reduces to

$$\frac{\Theta}{T_d} = \frac{1}{(J + GA)s^2 + (B + GC)s + (K + GD)}$$

In the ideal stabilizer, this transfer function will be very small for all disturbing frequencies. At low frequencies, the constant term in the denominator is important; near the ship's resonant frequency, the velocity term is important; and at higher frequencies, the acceleration term is important. If we are to accomplish any appreciable amount of stabilization over a wide range of frequencies, it is necessary that angular displacement and its two derivatives all be measured and used as control signals.

B. Dynamic Similarity in the Models

If a valid extrapolation of model test data to full-scale performance is to be expected, it is necessary that dynamic similarity between ship and model be achieved. This also implies static similarity. The hull form must be geometrically similar. If the ratio of ship length to model length is λ , the model must be loaded to a draft of λ^{-1} times the ship's draft. Its displaced volume must be λ^{-3} times that of the ship. Its displacement (weight) must be $\frac{\rho_f}{\rho_s} \lambda^{-3}$ times that of the ship, where ρ_f amd ρ_s are the densities of fresh and salt water, respectively. Its vertical and horizontal centers of gravity, its center of buoyancy, and its metacentric height must all be λ^{-1} times those of the ship. Its moment of inertia about the axis of roll, which can be expressed as a product of mass times radius

of gyration squared, must be $\frac{\rho_f}{\rho_s} \lambda^{-5}$ times that of the ship. The foregoing relationships will satisfy static similarity requirements.

To insure dynamic similitude, conventional towing-tank procedures will apply. Ship and model must be operated at equal Froude numbers (equal speed-length ratios). This requires that $V_m = \lambda^{-\frac{1}{2}} V_s$, where V_m and V_s are model and ship velocities respectively. It is normally impracticable to maintain equal Reynolds numbers in ship and model. One might ask whether the Reynolds number would not be particularly important with regard to the fins, which operate where viscous effects predominate. Fortunately, Reynolds number has negligible effect on the behavior of similar fins, other than to change slightly the angle of attack at which breakdown of flow occurs,and to produce a negligible change in the drag coefficient. As an example of the order of magnitude of the breakdown phenomenon, two fins, one ten feet long and the other $1\frac{1}{2}$ inches long, will experience breakdown at about 35° and 30° angle of attack respectively, when operated at the same Froude number. It is not expected that this will seriously impair the efficiency of the smaller fin.

An aspect of dynamic similitude which is becoming increasingly important in scale model testing is the passage of time. When towed at a Froude number equal to that of the ship, events occur $\lambda^{\frac{1}{2}}$ times faster for the model. If the ship's natural period of roll is N seconds, that of the model

must be $\lambda^{-\frac{1}{2}}$ N. It is important to remember this in the design of the stabilizer servo. If a system has proved to be satisfactory in the model, its time constants may be $\lambda^{\frac{1}{2}}$ greater in the ship and the same performance could still be expected.

Knowing the scale factors for length, mass (weight), and time, any other scale relationships between the ship and model may be determined by dimensional analysis. For example, if the power output of a model servo motor is known, we may wish to determine an equivalent full-scale motor. Power has the dimensions of (force) (length) $(\text{time})^{-1}$ or equally (mass) $(\text{length})^2$ $(\text{time})^{-3}$. Substitution of scale relationships for these dimensions indicates that the power of the larger motor must be $\lambda^{7/2}$ times larger than the model motor, ignoring the scale effects of the water densities.

C. <u>Physical Requirements of Model Equipment</u>

Before designing a servomechanism for use in a towed ship model, it is necessary to determine the limitations in size, weight and type of equipment which may be used. The authors have designed a system for use in a $5\frac{1}{2}$-foot destroyer model. This is a particularly difficult application, since the small size of the model places severe restrictions on the weight and size of the equipment located in the model. Further, its period of roll is shorter than that for most large ship models.

The model must displace about 20 pounds. The weight of the wood in the bare hull is about six pounds, leaving about 14 pounds to be loaded as equipment and ballast. This small weight allowance dictates that the servomechanism should be electric rather than hydraulic. It is also necessary to lead power and control connections to the model. This is most easily accomplished in an electric system.

The model restricts any equipment located within it to a width of about $5\frac{1}{2}$ inches. It is also necessary that the center of gravity of all equipment be about $3\frac{1}{2}$ inches above the base line, in order to achieve static similarity.

With these restrictions in mind, the authors chose the equipment which will be described in subsequent sections. Time did not permit installation and operation of the equipment in a model in a towing tank. It was, however, operated in an analogous system on the laboratory bench, and the authors feel that with certain modifications, which will be mentioned later, the system should be perfectly satisfactory in a model.

Although the model could probably have been loaded to the proper displacement and trim, it would have been rather difficult to achieve the proper inertia and metacentric height. The total weight of bare model and equipment was nearly equal to the required displacement, leaving very little margin with which to adjust radius of gyration and center of gravity. A model somewhat larger than the model previously described would present less difficulty in this respect.

In a towing tank such as that at M.I.T., where the model is not propelled by a moving carriage overhead, leading the numerous electrical connections to the model presents a problem. It is desirable to have the amplifiers and other auxiliary equipment as close as possible to the equipment in the model in order to minimize line drops and radiation. The wiring should not add to the weight of the model, nor interfere with rolling or towing. The authors suggest that all of the auxiliary equipment be placed in a suitable cart, which may be wheeled alongside the model as it moves down the tank. A "fishing pole" arrangement may be used to carry the electrical leads to the towing bracket. The leads should fall freely into the model. The towing bracket should be hinged longitudinally in the model to permit rolling. It would be necessary to trail only the primary 60 cycle and 400 cycle power leads from the cart.

In tanks which are not equipped to create waves from the model's beam, the roll must be artificially induced. The authors suggest that this might be accomplished by means of a moving weight. This weight would be on a rotating arm being driven by an electric motor mounted with its axis vertical. Both the static moment arm of the off-center weight, and the dynamic effect of the centrifugal force of the weight, would contribute to the roll of the model. It would be necessary to calculate the weight and arm roughly for each application, and the amplitude of roll might be varied by sliding the weight in or out on its arm.

The authors suggest that a 400 cycle servomotor with attached tachometer would be well suited for this purpose. A suitable amplifier circuit to be used with a Kearfott R-300 motor is given in Figure 2. This motor, using a tachometer feedback, offers the advantages that the frequency of the roll can be varied, the changing torque load on the motor would not materially affect its speed, and it is lighter than a d.c. motor. It is believed that this method of artificially creating roll would produce a reasonable facsimile of natural roll.

III.

THE SERVOMECHANISM

A. General Outline

Before proceeding to a detailed description of the apparatus, let us examine the system in broad outline. The equipment may be categorized in two groups: the sensing elements and the control elements. The sensing elements measure the angular displacement, rate and acceleration of the model. These consist of a stable platform which indicates the angle of roll and which can be modified to indicate the rate of roll, and a linear accelerometer. The entire system operates on 400 cycle power. The accelerometer signal is a d.c. signal, and must be made to modulate a 400 cycle carrier wave. These three signals are fed to a summing amplifier in which their magnitudes can be independently varied. The output of this amplifier is the primary control signal.

The servomotor driven by the control signal is a 400 cycle motor with attached tachometer. It drives the fins through reduction gears, and is itself stabilized by means of position and rate feedback. The lift of the fins, acting through the fin moment arm, produces a torque on the model in opposition to any disturbance. The block diagram of this system is that previously referred to in Figure 1.

B. The Electromechanical Analog

No matter how well-founded may be the theory in a problem of this sort, one is inevitably beset with a multitude of practical problems. There is many a slip 'twixt paper and hardware. Saturation and phasing troubles are encountered in the electronic components. Oscillations occur at the drop of a hat, and loop gains must be set accordingly. It is highly desirable that the electrical system be made to operate satisfactorily before installation is made in the model, if for no other reason than that the possibility of instability and consequent capsizing of the model be minimized.

We, therefore, need some means of simulating model operation on the laboratory bench, utilizing as many of the final components as possible. The authors have built an electromechanical analog system which electrically simulates all of the torques, but which otherwise employs all of the elements of the model system.

The model is duplicated by a swinging platform driven by a 400 cycle servomotor through a gear train. This is shown photographed in Figure 3. The stable element is mounted in the platform, with its own sensitive axis aligned with the platform axis. The accelerometer is secured to an extension of the platform. The platform must be rigid, there must be a minimum of shaft lengths in the gear train, and gear teeth must be carefully meshed, all in order to eliminate mechanical vibration and backlash.

The motor, a Kearfott Model R-801-1A-A, is driven from an amplifier circuit which is shown in Figure 4. With this arrangement, it is possible to duplicate the dynamic performance of any ship model desired. The platform, gears, and motor provide the mass of the system. Although some friction is present, any desired amount of damping may be obtained by adjusting the tachometer feedback loop gain. The necessary degree of stiffness is regulated by varying the synchro feedback loop gain. The block diagram for this system is shown in Figure 6, which is seen to represent the equation of a ship model whose parameters may be adjusted. The schematic wiring diagram for the amplifier-motor circuit is reproduced in Figure 5.

A ten-turn Helipot is used to represent electrically the hydrodynamic effect of the fins. The shaft and center-tap of the Helipot are driven through gearing by the fin servomotor. The outside terminals of the Helipot are connected to the outside terminals of a three-tap transformer secondary winding, as shown schematically in Figure 7. The transformer center-tap is grounded, so that the voltage at the Helipot center-tap is a direct measure of the angle of tilt of the fins, the 400 cycle carrier being modulated by positive and negative position angles. This arrangement, however, does not reproduce the lead effect of the fins due to their transverse motion through the water.[5] In that

respect, the model in water would be expected to perform better than the analog.

The Helipot output voltage, which is considered to be directly proportional to the torque of the fins, is fed back into the amplifier-motor representing the ship model, thereby simulating the stabilizing moment of the fins.

Wave disturbance is produced electrically by using the stator voltage of a motor-driven synchro whose motor is excited by 400 cycle power. This sine source disturbance, which may be varied in both amplitude and modulation frequency, is the primary input to the ship model amplifier. It is shown photographed in Figure 8.

The entire analog system is shown schematically in Figure 9, and is pictured in Figures 10 and 11. All components which would be used in the model are used in the analog, with the exception of the fins and the model itself, both of which must be operated in a fluid. Using this scheme, any type of ship model may be represented, and any configuration of sensing and control elements may be tested and adjusted.

C. The Sensing Elements

The heart of the sensing components is a stable element about which the model rolls. This is a single-axis stable platform carrying a HIG-4 integrating gyro. Its electronic components are shown schematically in Figure 12. The platform is maintained stationary in space by a servomotor which

is geared to the foundation. A synchro is also geared between the foundation and the platform, and the voltage of one of the stator phases is a direct measure of the angular displacement of the foundation and of the model. It was found that within the range of frequencies of interest, the stable element could be considered to have no appreciable lags. At frequencies up to about six cycles per second, no motion of the platform could be detected. The authors, being largely ignorant of gyro techniques, are happy to label the entire platform as a single gain constant between roll angle and synchro voltage.

A small amount of drift due to the earth's rotation does occur using this type of platform. In its orientation on the laboratory bench, this amounted to less than a degree per minute, and under the worst circumstances, four degrees drift per minute would be observed. It is believed that in most cases, drift will not seriously affect the validity of the tank tests.

At the outset of the experimental work, it was hoped that a measure of angular velocity could be obtained from the control voltage on the platform's servomotor. This proved to be impracticable, due largely to the amount of noise present in the signal. For purposes of laboratory operation, the velocity signal was obtained from the tachometer attached to the motor driving the simulated model.

This is admittedly a devious means of obtaining the information, and would of course not be possible in model operation. However, it served the very useful purpose of demonstrating the behavior of the servo had the rate signal been readily available. In a later section, a recommended modification to the platform will be given, whereby a measure of angular velocity may be easily obtained.

A Calidyne Model 18 D-5 linear accelerometer was employed to measure angular accelerations. The accelerometer is mounted at some distance (perhaps a foot) above the axis of roll, with its sensitive axis tangent to the roll. The linear accelerations which it measures are then proportional to angular accelerations.

This type of accelerometer consists of a cantilevered mass acting through a diaphragm to move the plate of a vacuum tube. A schematic of the circuit is shown in Figure 13. A d.c. voltage is applied across the tube, and the variable contact of a potentiometer is used to establish the point of zero potential with no acceleration applied. Acceleration of the unit serves to change the plate resistance of the tube and hence to change the voltage at the potentiometer pick-off. This d.c. voltage is applied to an Airpax Model A-580, 400 cycle mechanical chopper. The output voltage on the vibrating reed member is a 400 cycle square wave modulated by the accelerometer signal. The chopper output passes

through a capacitance which removes the d.c. component of the signal and which passes the square wave as a suppressed carrier signal.

The sensitivity of the Calidyne Model 18 D-5 accelerometer is 5 volts/g acceleration with the type of output circuit used. The accelerations experienced by the model's analog are no greater than about 0.2 g, so that the data voltage is rarely greater than one volt. This is a considerably lower potential than that carrying the position and rate information. In addition, the noise in the signal, due largely to radiation pick-up, is of the same order of magnitude as the data, in spite of shielding. As an expedient measure, the acceleration signal was amplified through a Ballantine Model 643 a.c. voltmeter before going to the control system. This voltmeter may be used as an amplifier, but saturation occurs at about 30 volts on the output stage. The measurement of acceleration is, in short, a continuing problem, and one whose solution is vital to an optimum stabilizing system. It is discussed further in Chapter IV.

D. The Control Elements

The elements of the control system comprise an electronic summing amplifier, a motor with attached tachometer, and the Helipot which was previously mentioned as being analogous to fins. The amplifier is shown schematically in Figure 14, and is suitable for 57.5 volt (parallel) operation of the servo-

meter. The unusual number of inputs are required to accommodate the three data signals from the sensing elements in addition to the feedback inputs.

The motor-tachometer is a Kearfott Model R-800-1A-A. This motor is suitable for use in model operation, and its selection was based on the calculated torque loads which it would be required to overcome. The electromagnetic motor torque is opposed by:

1) Hydrodynamic load on the fins
2) Friction
3) Inertia

The hydrodynamic load is that torque which is exerted on the fin shafts by the lift and drag of the fins. Calculations were made for two fins of 1.681 inches span, 0.756 inches chord, and 0.161 inches thickness. A maximum tilt of 23° was assumed, and to this figure must be added the increased angle of attack due to lateral motion of the fins during rolling. The fin shaft was assumed to be at 0.296 chord length. Calculations were made using Joessel's standard rudder formulae, and were confirmed by Schoenherr calculations[6]. A maximum torque of 0.51 ounce-inches per fin can be expected from hydrodynamic effects.

Friction occurs in the viscous fluid, in the fin shaft sealing glands, and in the bearings.

The viscous resistance encountered by the fin rotating in the water was calculated by assuming a fluid impingement

normal to the surface of the fins as they rotate. Using Reference (7), page 279, this effect was found to be negligible.

A crude estimate of friction torques in the sealing glands was made by assuming a pressure of 2 psi between gland and shaft, a 3/16 inch shaft, a seal 1/2 inch long, and a coefficient of friction of 0.5. A torque equal to 0.44 ounce-inches from this source was estimated. Bearing friction is believed to be small in relation to this value and is neglected.

In calculating inertia torques, it was assumed that the major load inertia would be due to the larger gear between motor and fins. As an estimate, a two inch brass gear, 1/4 inch thick, experiencing an acceleration of 250 rad/sec^2, would exert a torque of 0.21 ounce-inches.

Under the worst circumstances, all of these torque loads could be assumed to act together. They total 1.16 ounce-inches torque per fin, or 2.32 ounce-inches for two fins. To allow for error it was required that 3 ounce-inches torque be delivered to the load.

It is next necessary to strike a balance between motor torque and load torque, and between motor speed and load speed, with the gear reduction ratio as the conversion factor. It may be expected that the maximum rolling frequency of the model might be three cycles per second, for which the fins might reasonably be required to travel their

full throw in 0.1 second at an average velocity of 2 cps. Reference to torque-speed curves for the Kearfott Model R-800 motor shows that for medium control excitations, this motor delivers about 0.4 ounce-inches torque at a speed of about 20 cps. If a gear reduction ratio of about 10 is assumed, the inertia load of the motor itself is computed to be about 0.1 ounce-inches, leaving 0.3 ounce-inches torque to be delivered to the shaft. The ratio of this torque to the load torque is about one to ten, which is the inverse of the required speed ratio. This value for gear reduction ratio appears to be satisfactory.

Having selected the proper motor for the application, it is next necessary to insure its optimum response in the control system. The use of tachometer feedback with a high loop gain serves to reduce the motor's time constant to the point where its lag does not seriously impair system performance. Chadwick(5) suggests that this remaining lag may be cancelled by the inherent lead of the fins operating in water.

The analog fins were observed to have a strong tendency to drift from zero angle of tilt. This is due to stray unidirectional signals entering the motor amplifier, and to unbalance in the amplifier circuit. Use of the Helipot output voltage as a position feedback loop serves to eliminate this source of trouble. Even though the Helipot was introduced into the analog primarily to simulate fin moment, there

appears to be no reason why such an arrangement could not be used in a model to stabilize the fin servo loop.

The entire control system is shown schematically in Figure 15. It can be seen that the position and rate loop gains may be varied to attain almost any dynamic characteristics of the control servo, and the relative magnitudes of the three data signals may be varied to obtain optimum performance of the entire system.

E. System Performance

Having the individual components of the system in mind, the system as a whole may now be discussed. A block diagram of the entire analog is shown in Figure 16. Two inferences may be drawn from this diagram. The first is that high gains in the major feedback loop will reduce the overall gain of the transfer function, thus reducing the roll due to disturbances. The second is that oscillations may occur in the major loop or the two minor loops if gain values are too high. In general, increasing the loop gains produces improved roll stabilization up to the point of incipient oscillation. This point of optimum stabilization is best arrived at experimentally.

The authors arrived at such an experimental optimization in the following manner. The feedback voltage from the Helipot to the torque-summing amplifier (ship model analog) represents the effect of fin moment on the model.

A variation of the feedback potentiometer for this voltage would represent a variation in fin size, model velocity, fin moment arm, or any other parameter affecting fin moment. A zero setting on this potentiometer simulates an unstabilized ship, or a stabilized ship with the fins inoperative.

The proper dynamic characteristics of the model are established by adjustment of the tachometer and the synchro loop gains around the model. With the potentiometer controlling fin feedback set to zero, a disturbing voltage from the modulated sine source is put into the analog. The model begins to roll in response to the disturbance. The amplitude of roll is controlled by regulating the input potentiometer of the disturbing voltage, and the frequency of roll is controlled by regulating the motor speed of the sine source modulator. The characteristics of the roll are being measured by the sensing elements, and the fin control servo is operating in response to this data. However, the main loop is not closed because fin torque voltage is not being fed back.

Raising the Helipot feedback potentiometer is equivalent to activating the fins. Upon activation, it may be found that stabilization is very effective, or that oscillations occur before stabilization becomes significant. In the latter case, it is necessary to adjust the relative magnitudes of the position, rate, and acceleration signals to the control circuit, and to adjust the overall loop gain. These settings vary with the frequency and amplitude of roll.

It will be found that one particular combination of gain settings produces best results for a given type of roll. With this combination established, it is then possible to open and close the fin feedback loop, and thus to compare the unstabilized roll of the analog with the optimum stabilization performance.

Such a comparison is shown in the photograph of Figure 17. These are oscilloscope photographs of the voltage of the stable platform synchro, which voltage is a direct measure of angular displacement. The large excursions represent the unstabilized roll of the analog, and the small excursions represent the stabilized roll after the fin feedback loop has been closed. It will be observed that stabilization has reduced the roll amplitude by about 90%. The roll frequency in this case was very near the resonant frequency of the analog, where stabilization can be expected to be most effective. The resonant frequency is not only the frequency where stabilization is most desirable, but also where it is easiest.

It was found that closing the feedback loop suddenly introduced sufficient transient effects to start oscillation in the system. It is necessary to raise the feedback gain gradually over a period of perhaps a cycle or two. This is not necessarily deleterious; it is in fact doubtful whether amplitudes would be damped out much faster were it possible to close the loop instantaneously.

In the reverse case, going from the stabilized to the unstabilized condition, the loop may be opened suddenly. This reverse process is shown in the lower part of the photograph of Figure 17. It will be noticed that the change occurs somewhat faster and more smoothly than in the previous photograph.

IV.

RECOMMENDATIONS FOR FUTURE WORK

A. General Comments

The roll stabilizing system presented in this thesis should not be regarded as a complete system for installation in a ship model. Time limitations, and non-availability of certain desirable system components, made it necessary for the authors to design a system, (1) using components which were readily available, and (2) incorporating a method of obtaining a rate of roll signal, which would not be available when the system was installed in a ship model. As has been mentioned previously, the system, as designed, was not installed in a ship model and tested; rather, the ship response, and the effects of fin torque were simulated electrically in the laboratory. For full description of the stabilizing system and the simulated ship system, the reader is referred to Chapter III. The authors contend that this system proves the feasibility and practicability of the use of model size, roll stabilizing systems in conjunction with the study and design of full scale systems. It is the aim of this chapter to point out the limitations of the system as designed, to make recommendations for improvement of the system, and to outline a few of the items requiring future work.

B. System Limitations and Suggested Modifications

In order for any roll stabilizing system to perform properly, over the complete range of roll frequencies normally encountered at sea, information regarding: (1) angle of roll, (2) rate of roll, and (3) acceleration of angle of roll, must be available for use as inputs to the system and for loop stabilization within the system. It is generally agreed that the relative importance of the various inputs varies with the roll frequency as follows:

1.) Low frequencies - angle of roll (θ),

2.) Resonant frequencies - rate of angle of roll ($\dot{\theta}$),

3.) High frequencies - acceleration of angle of roll ($\ddot{\theta}$).

For any given roll frequency there is an optimum combination of θ, $\dot{\theta}$, and $\ddot{\theta}$ which will give best stabilization of roll. For the system block diagrams and analysis thereof, the reader is referred to Chapter I, and to Chadwick[1]. Suffice it to say here that overall loop stability was the determining factor on the degree of roll stabilization achieved. Loop stability can be greatly improved by acceleration and rate feedback. The effect of such feedback is to raise the resonant frequency of the overall loop above that of the resonant frequency of the

ship, thus enabling the use of a higher closed loop gain, which in turn permits a greater reduction in amplitude of ship roll.

With respect to the foregoing discussion, there were two major limitations imposed on the system as described herein - the gyro, and the accelerometer. A discussion of the gyro and accelerometer and recommendations for improvement of the utilization of each is given below.

1. <u>Gyro</u>

The gyro actually used as the sensing device for roll was very adequate with respect to measurement of angle of roll (θ). The frequency response of the stabilized gyro platform was flat to above 5 - 6 cycles/sec and showed no attenuation over the frequency range required for model testing. Originally the authors had hoped to use the control signal of the gyro stabilized platform drive motor as a measure of the rate of roll of the ship. However, this control signal proved unusable since it incorporated both position (θ), and rate ($\ddot{\theta}$) feedback signals in order to establish a tight, stable closed loop around the gyro. After several further attempts were made to obtain a usuable rate signal from within the gyro control amplifier, it was decided to utilize the tachometer signal from the motor - generator used to drive the simulated ship platform. Even though this method yields a true rate signal of the ship's motion, it

must be emphasized that this method is available only in the laboratory and cannot be used when the system is installed in the model.

In order to eliminate the problem of obtaining a signal proportional to rate of roll ($\dot{\theta}$), it is recommended,

1.) A gyro be obtained which is capable of providing a rate signal as well as a position signal, or

2.) The existing gyro be modified by replacing the gyro stabilized platform drive motor with a motor-generator (tachometer) combination. The output of the tachometer will be directly proportional to the rate of roll ($\dot{\theta}$).

There are several other characteristics of the gyro which must be considered when selecting a gyro for use in a model installation. The entire gyro unit, sensing element and platform, must be as light in weight as possible. The gyro is the heaviest, single system component, and it is not inconceivable that the weight of the stabilizing system may exceed the permissible loading of the model to be tested. It is estimated that the total weight of the system presented in this thesis, including necessary foundations for mounting equipment in the model, would be approximately fourteen (14) pounds.

One of the most undesirable characteristics of any gyro is gyro drift. It is almost impossible to completely eliminate

this drift; however, low drift gyros are available commercially, and are desirable for use in model testing - especially where total time for conducting a model run is appreciable. Drift is detrimental from the standpoint of obtaining accurate results since there will be a discrepancy between the true vertical and the vertical established by the gyro. The ship model tends to seek the true vertical, whereas the model stabilizing system attempts to drive the ship to the gyro vertical. An appreciable amount of drift will therefore cause the ship to be stabilized about some point off the true vertical.

One last feature which would be desirable to have incorporated in the gyro is a means for accurately and remotely aligning the gyro vertical with the ship (or true) vertical at the start of each run. This is sometimes referred to as "zero setting". One method of "zero setting" is discussed by Porter.(8)

2. Accelerometer

The accelerometer used by the authors was a Calidyne Model 18D-5, which has a range of 0-5 g with a sensitivity of 5 volts/g. This accelerometer, which measures linear acceleration, was mounted approximately one (1) foot from the center of roll. An average value of acceleration actually experienced by the accelerometer was less than 0.2 g, which produces a d-c output of the order of one (1) volt. In order

for the accelerometer signal to the fin drive amplifier to be of the same order of magnitude as the rate ($\dot{\theta}$) and position (θ), a separate amplifier is required for the accelerometer output signal. The noise level of the accelerometer output signal was quite high and large amplification of this signal was virtually impossible.

Using this accelerometer in the model would pose several problems. The first problem is how to mount the accelerometer at a substantial distance from the axis of roll in order to take full advantage of the sensitivity and obtain a better signal-to-noise ratio. The second problem is to insure that the mounting bracket (or stand) for the accelerometer is essentially vibration free, and that any mechanical vibratory frequencies are well above the natural frequency of the accelerometer. This requires a rather "stiff" mounting, which is acquired only at the expense of additional weight.

In view of the existing problems with the use of this accelerometer, the following recommendations are made:

1.) Use of an accelerometer with a similar low frequency response, but with a greater sensitivity. It must be remembered, however, that with increased sensitivity, extreme care must be exercised in the elimination of extraneous vibrations of the ship model and accelerometer mounting.

2.) Use of an angular accelerometer, in lieu of the linear type used herein. The angular type accelerometer could be mounted on the axis of roll, and therefore would not require the special off-center mounting of the linear type. Of equal importance is the reduced effects of coupling from accelerations in the other five degrees of ship motion into the roll degree of freedom.

3.) Regardless of the type of accelerometer employed as a sensing device, the complete electronic circuitry for the accelerometer must be shielded. This is especially true where all wiring between the model installation and the shore-based components must, of necessity, be run through the same conduits and thus be in close proximity to each other. The authors experienced some difficulty with interaction of the gyro heater circuit (400 cps) with the accelerometer circuit. Shielding reduced this interaction considerably. Further improvement of this situation could be realized by changing the gyro heater circuit from 400 cycles/sec to 60 cycles/sec. This is especially true if a 400 cycles/sec stabilizing system is being used.

3. Fin Drive System - Position Feedback

In order to minimize the error between actual fin position and ordered fin position, position feedback was employed to close the loop around the fin drive motor and

its associated amplifier. In the laboratory a Helipot was used to simulate fin position and lift. The output of the Helipot was used as a measure of fin position. For details of this installation, the reader is referred to Chapter III. In order to obtain this position signal when the system is installed in the model, it is suggested that a similar Helipot, driven by the fin shaft, be installed in the model. Without this position feedback, the fins will not return to the zero lift position when the ordered fin lift signal is zero.

C. Suggestions of Areas for Future Study

1. Analysis of Scaling Parameters

The ultimate aim of the use of ship stabilizing systems on a model scale is not merely to stabilize the ship model, but to be able to predict the performance of a full scale stabilizing system. This is done by extrapolation of model data to full scale predicted data. To the knowledge of the authors, no previous work has been done in this field. Before model data can be of substantial value to the designer of ship stabilizing systems, suitable scaling laws for system parameters must be developed, and correlation proved. A list of the more important parameters requiring scaling laws would include:

1.) System time constants,

2.) Installed torque and power capacity of fin drive system,

3.) Roll torque produced by the fins, including size and configuration of fins and ship's speed,

4.) Amplifier gains and loop stability criteria,

5.) Relative values of various input signals within the system,

6.) Conversion from electrical components to hydraulic components, including any limitations imposed on either system.

2. Damping Effects of Stabilized Gyro Platform

The authors made no attempt at assessing the effect of the stabilized gyro platform on the damping coefficient associated with ship's motion in roll. However, Allen[3] reports that a gyro recorder, when mounted in the model, produced definite friction damping. For more accurate results from model testing, the value of such damping should be known and properly taken into account when reporting results of model tests.

3. Other Types of Sensing Devices

The use of sensing devices to measure the ship motion which the stabilizing system is trying to eliminate leaves

much to be desired. A ship possesses considerable inertia and weight, and gains momentum rapidly when in motion. It would be desirable, therefore, to be able to measure the exciting force before it acts rather than the resultant motion due to this force. This would require some sort of sensing device to measure such characteristics of the approaching waves as height, frequency of encounter with the ship, and wave slope. Efforts have been made in this direction, but little progress has been made. It should be noted also that even if the exciting force could be defined, complications would still arise from the simple fact that different type ships will respond differently to a given wave.

4. Extension to Pitch Stabilization

The entire theory presented herein, as well as much of the equipment discussed, is applicable to the stabilization of pitch. This is a field which will be of increasing importance in connection with seaworthiness studies.

V.

SUMMARY AND CONCLUSIONS

The authors have presented what is essentially a first study of suitable hardware for use in stabilized ship models. They have attempted to persuade the reader that such model tests are potentially of considerable value, and that they may be necessary if complete optimization is to be achieved. The suggested servomechanism has been described in terms of its transfer function, its individual components, and its performance. An analog system for proving the sensing and control components before installation in the model has been suggested.

First efforts are rarely perfect, and this work has been no exception to the rule. Limitations of and faults in the equipment have been observed and pointed out. The authors have made recommendations for removal of these shortcomings, which should spell success in any future research.

The work described in this paper is only part of what must be accomplished, if credible stabilization test data is to be obtained from ship models. It will be necessary to improve the instrumentation and to install it in ship models. This will undoubtedly raise new problems. When satisfactory

operation in the towing tank is achieved, the next task will be to verify the relationships between model performance and ship performance. With the laws of similitude proven, it will then be possible to study any number of model configurations and thus to attain a truly optimum ship stabilization system.

A P P E N D I X

A. Figures

B. References

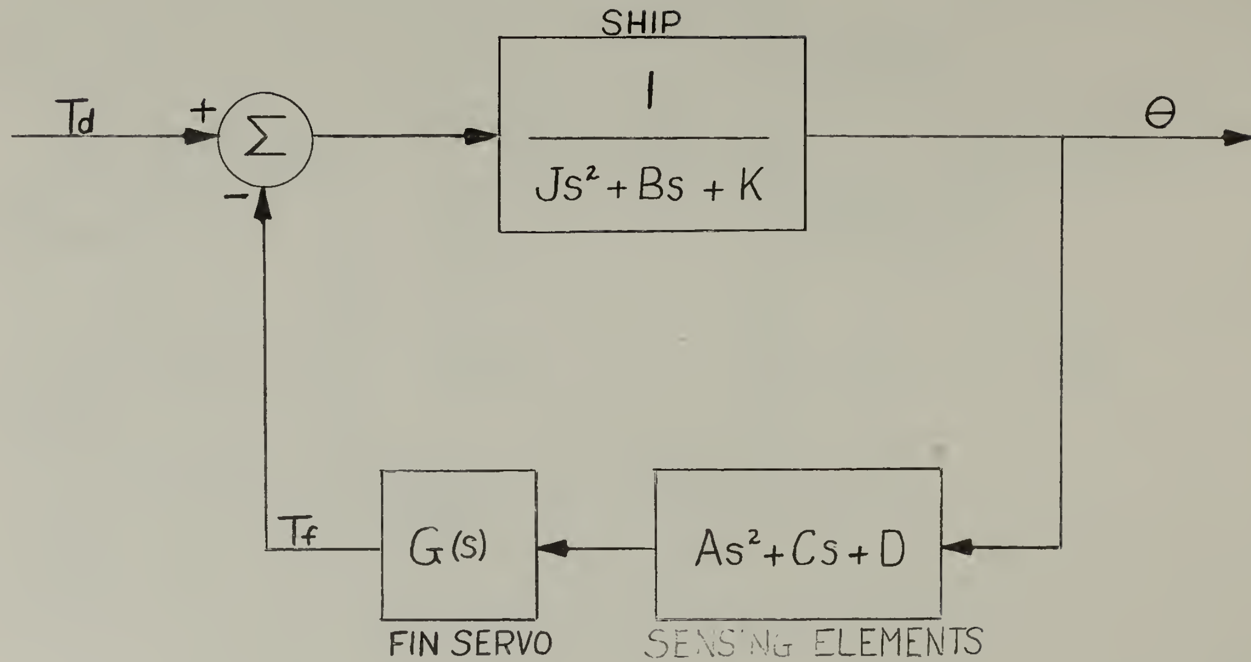


FIGURE I. STABILIZED SHIP
SIMPLE BLOCK DIAGRAM

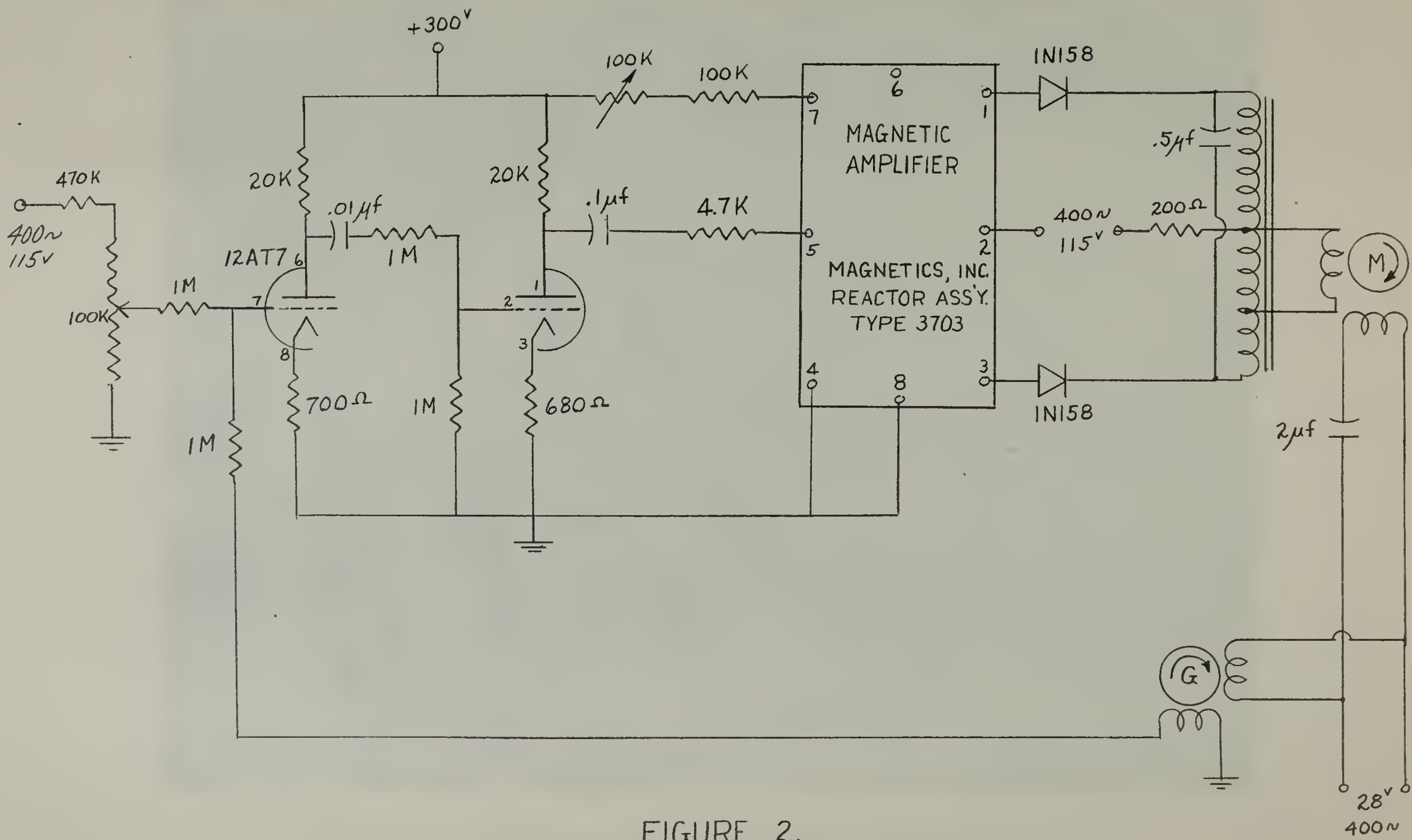


FIGURE 2.
ROLL MOTOR AMPLIFIER CIRCUIT

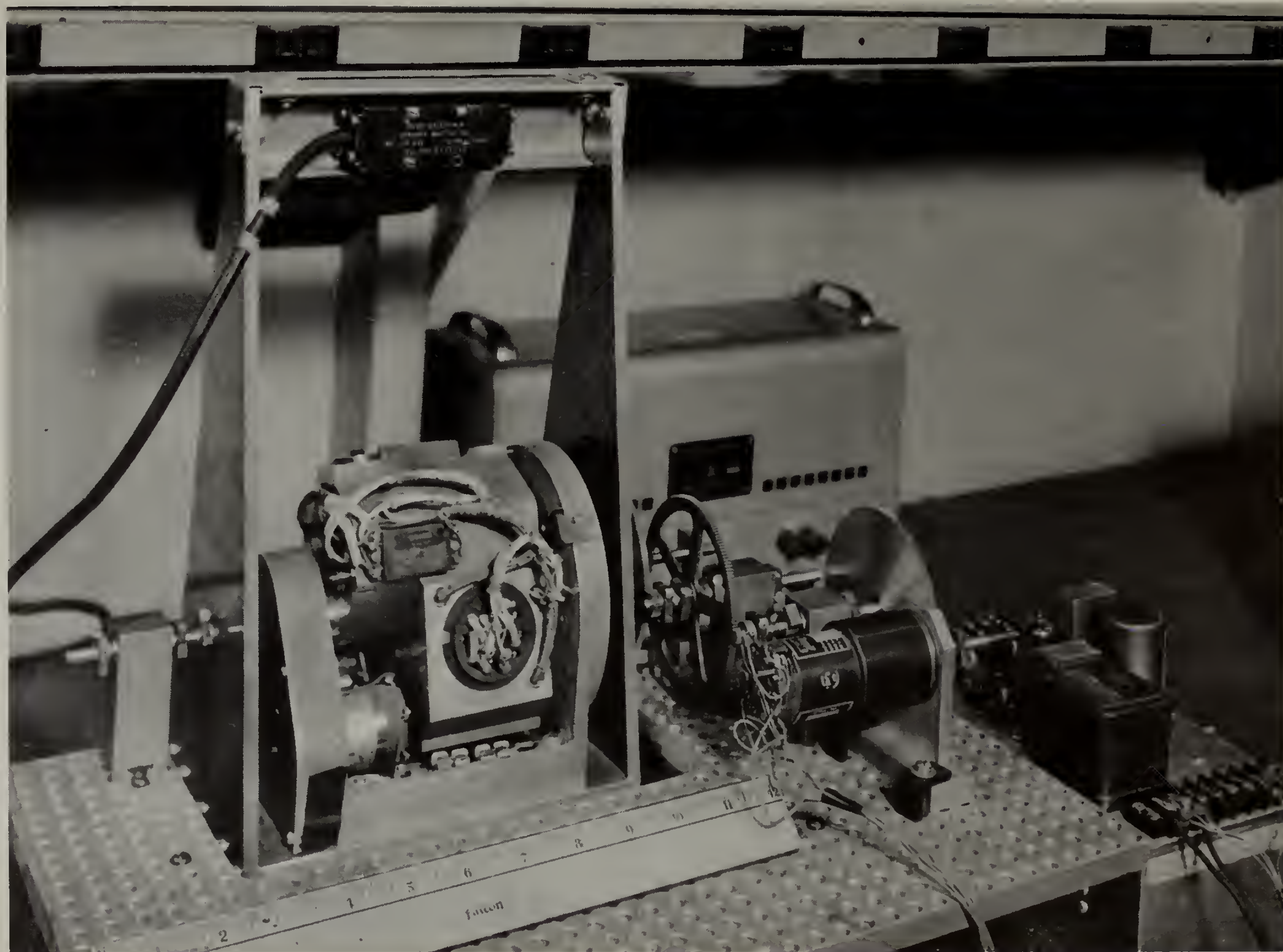


Figure 3. Swinging Platform Driven by Motor Used as Analog of Ship Model Among Waves.

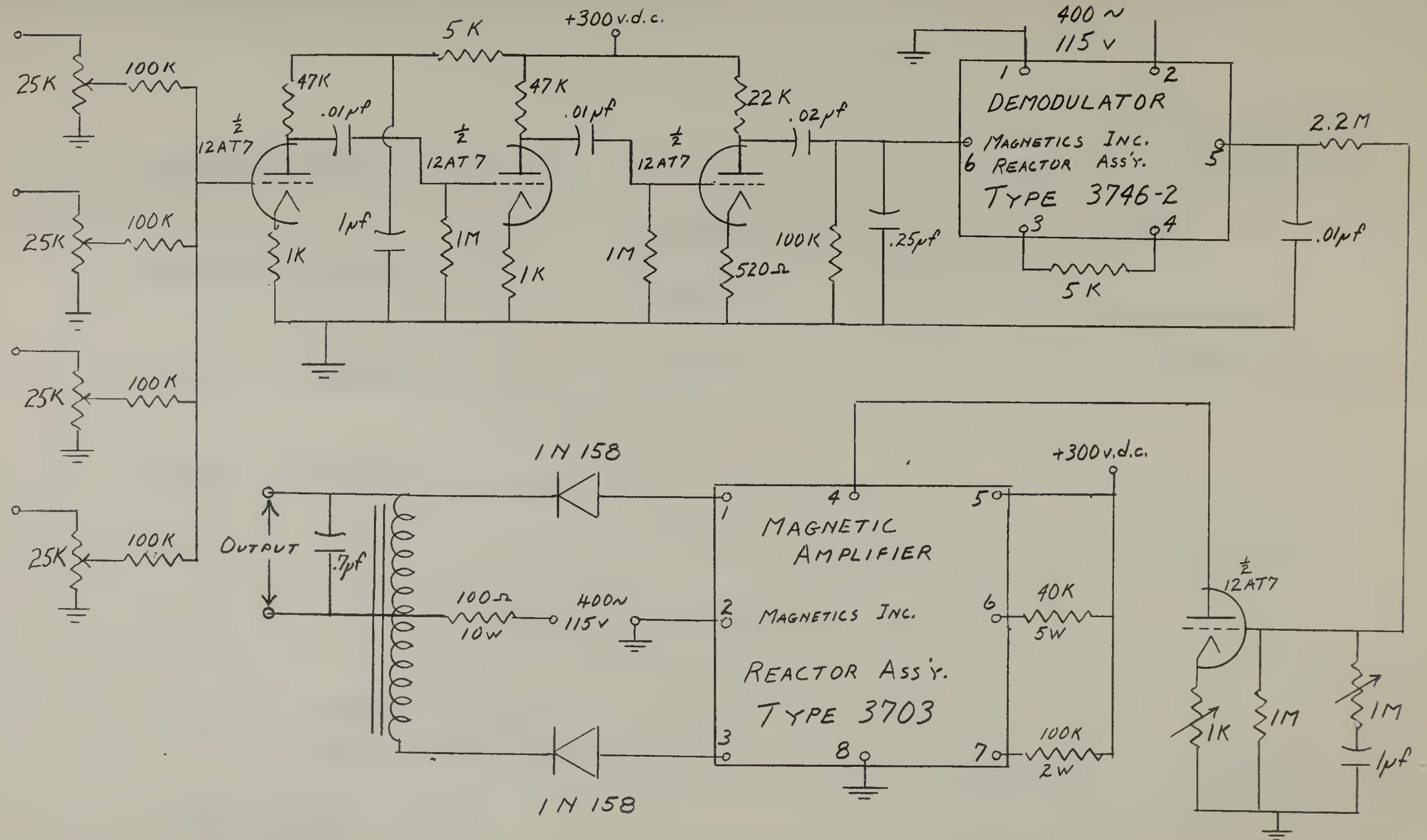
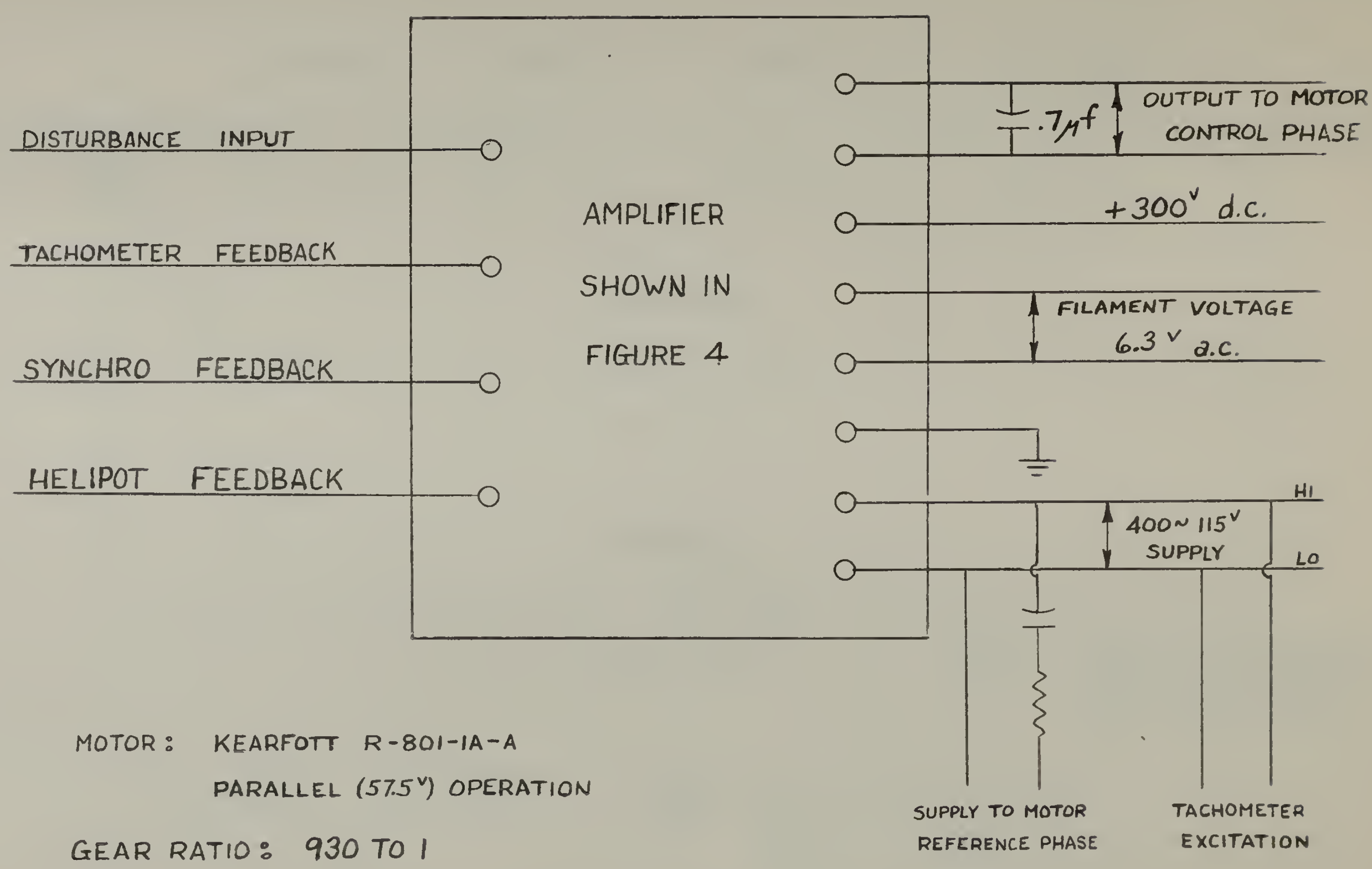


FIGURE 4. AMPLIFIER FOR MODEL ANALOG DRIVE MOTOR

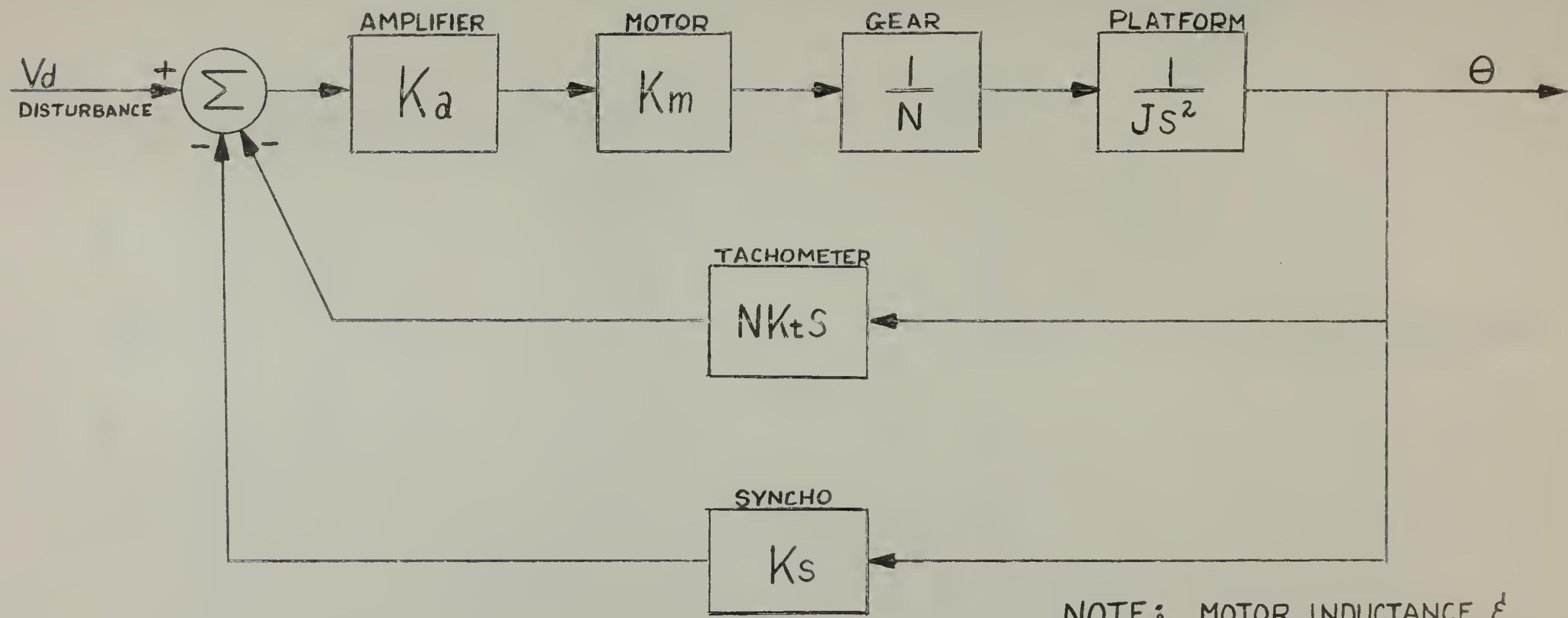


MOTOR: KEARFOTT R-801-1A-A

PARALLEL (57.5v) OPERATION

GEAR RATIO: 930 TO 1

FIGURE 5. SCHEMATIC WIRING DIAGRAM FOR MODEL ANALOG MOTOR AMPLIFIER

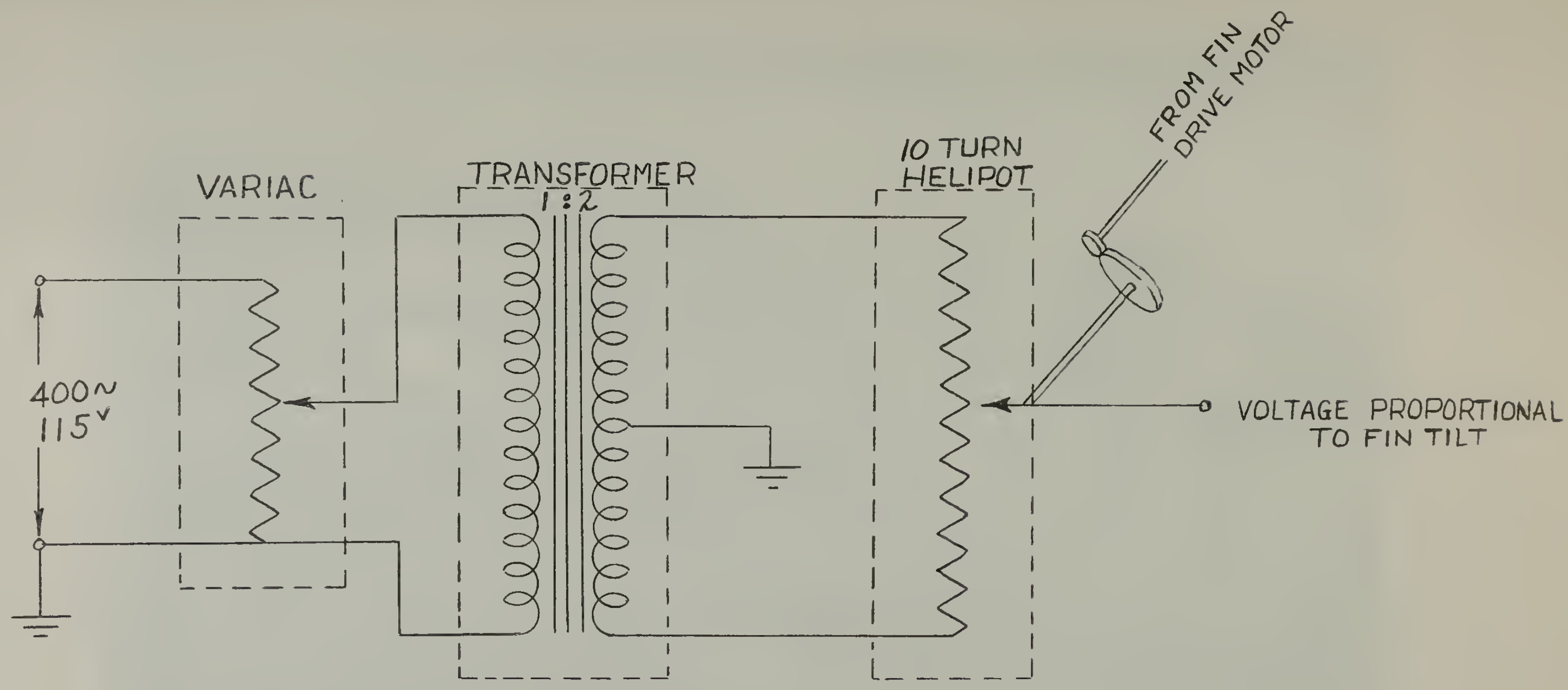


NOTE: MOTOR INDUCTANCE & MECHANICAL FRICTION ARE NEGLECTED.

TRANSFER FUNCTION:

$$\frac{\theta}{V_d} = \frac{1}{\frac{NJ}{K_a K_m} S^2 + K_t N S + K_s}$$

FIGURE 6. BLOCK DIAGRAM OF MODEL ANALOG



HELIPOT: MODEL A, 10 TURN
5 WATT, 1000-Ω

GEAR RATIO: 5 TO 1

FIGURE 7. SCHEMATIC WIRING DIAGRAM OF FIN TILT ANALOG

Figure 8. Source of Sinusoidally Modulated 400 Cycle Voltage Used as Analog of Wave Disturbance.

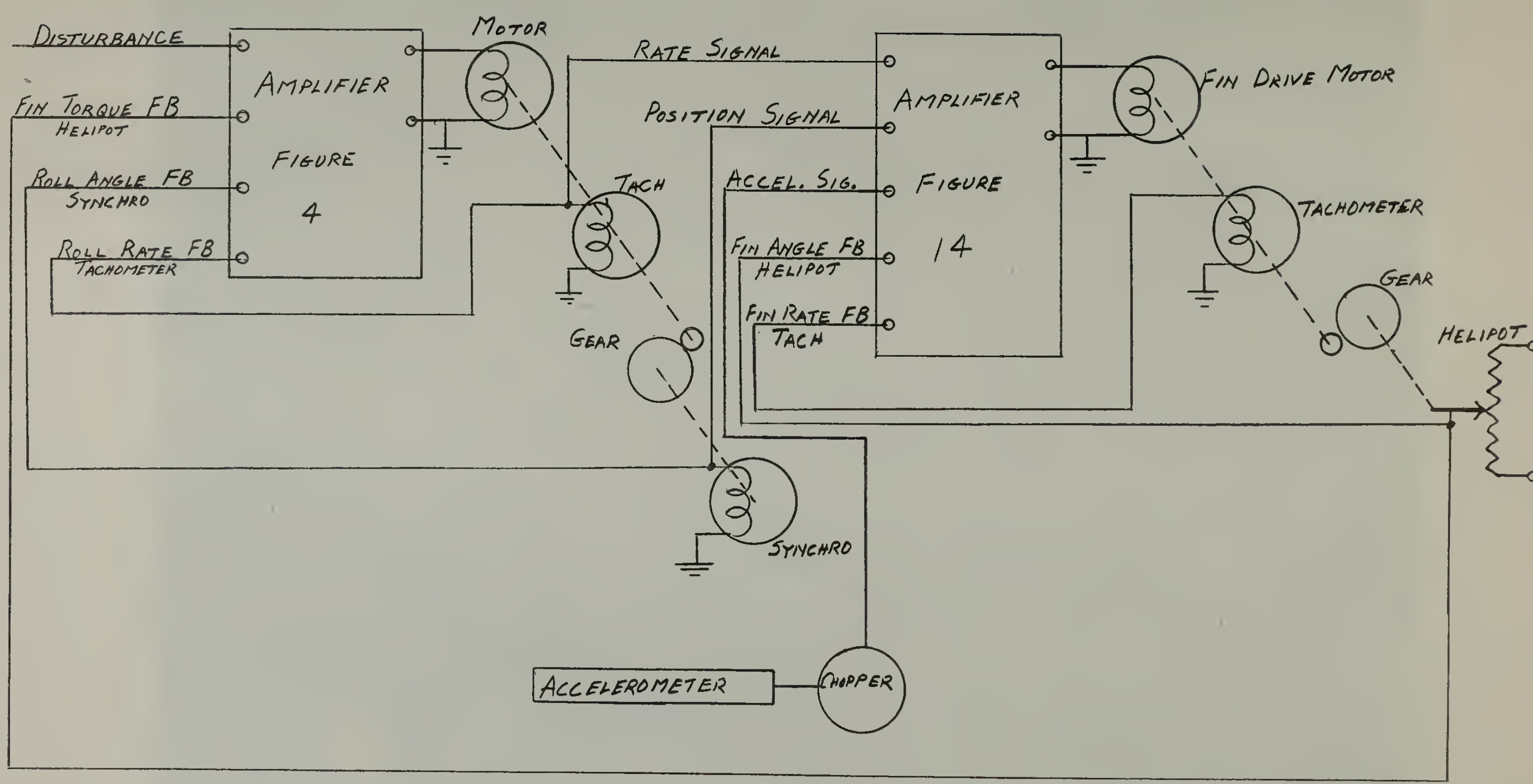


FIGURE 9. SCHEMATIC OF ENTIRE ANALOG SYSTEM

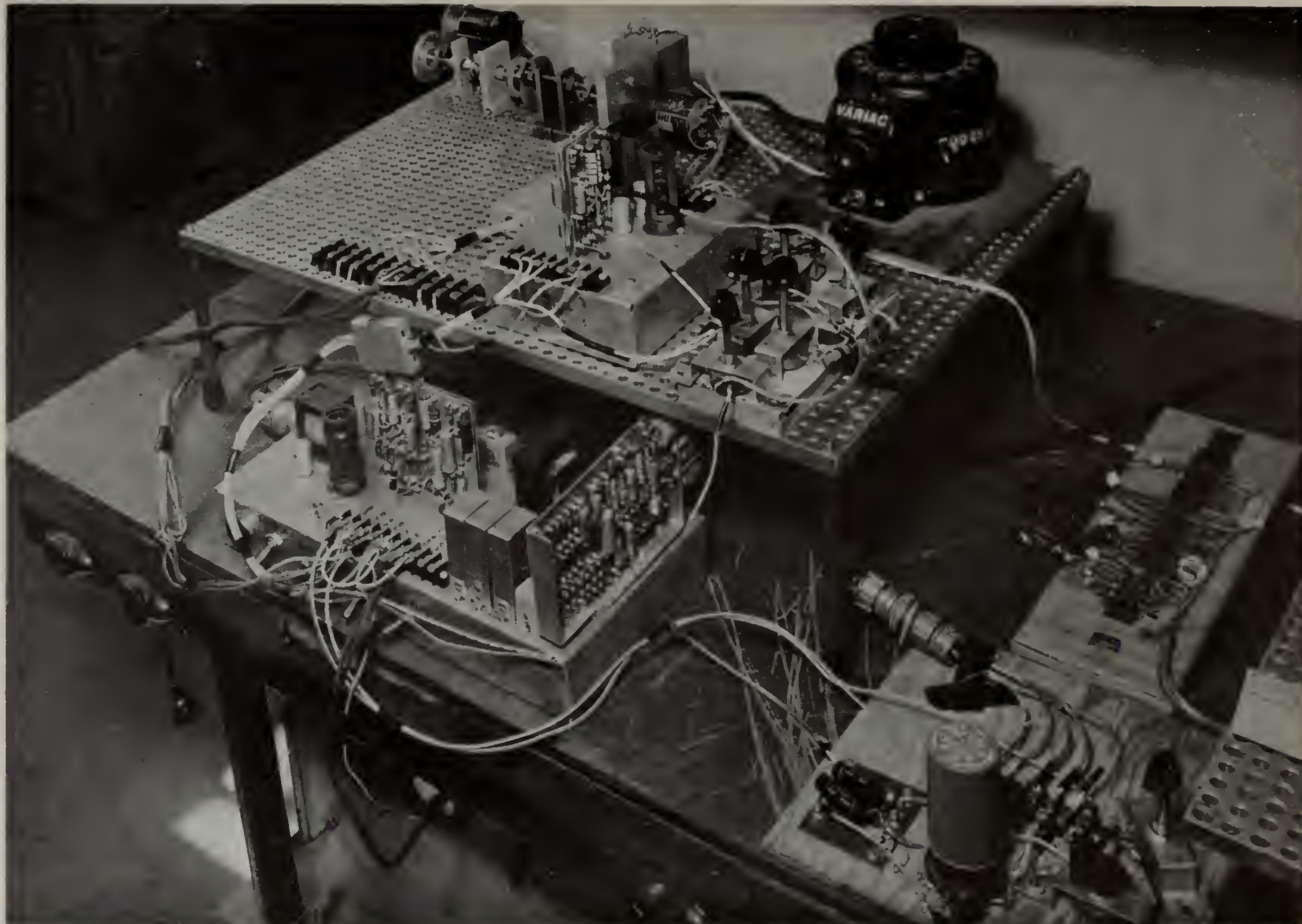


Figure 10. Fin Drive Motor and Gear, Including Transformer and Helipot Used as Analog of Fin Tilt and Fin Moment.

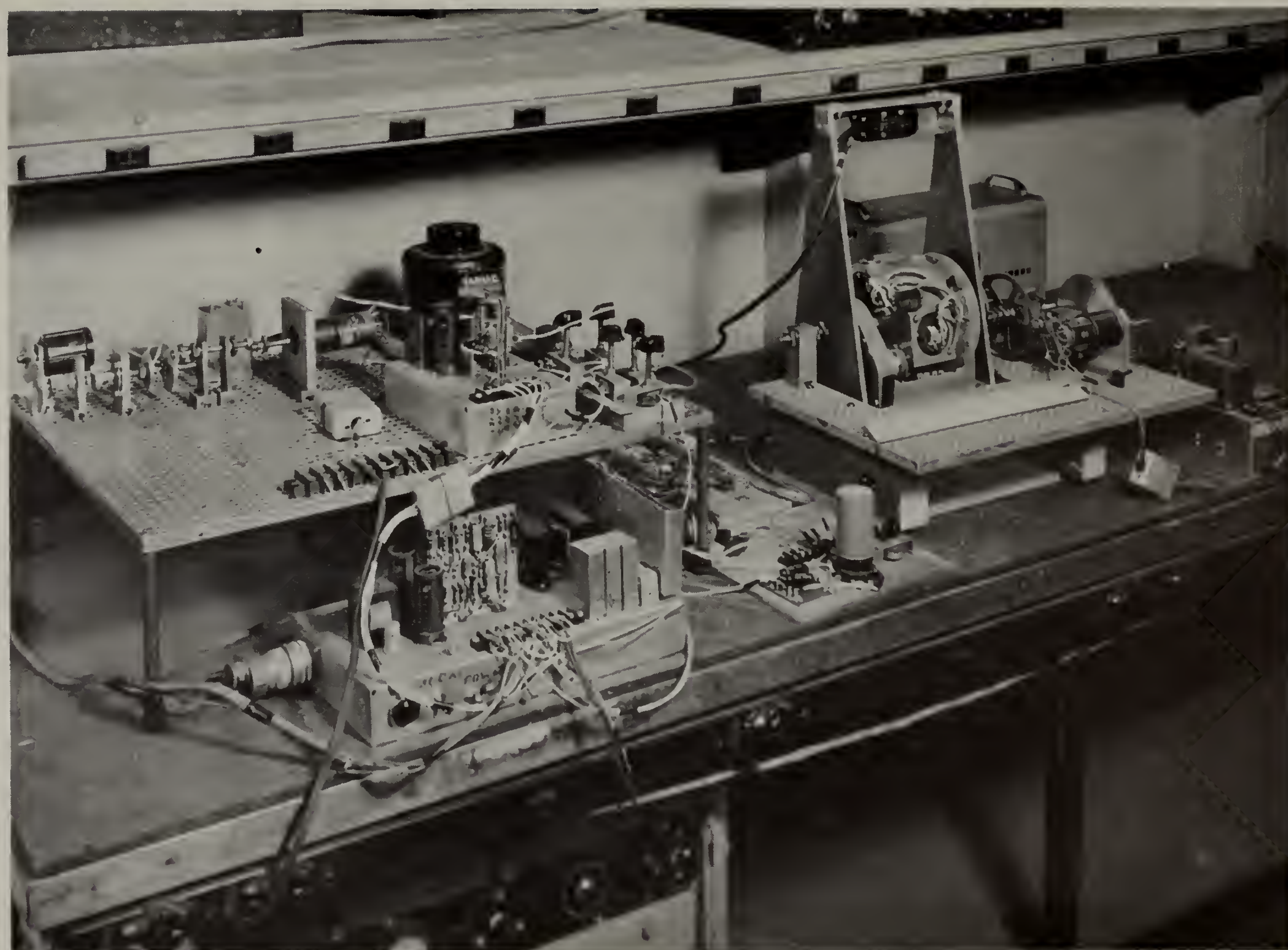
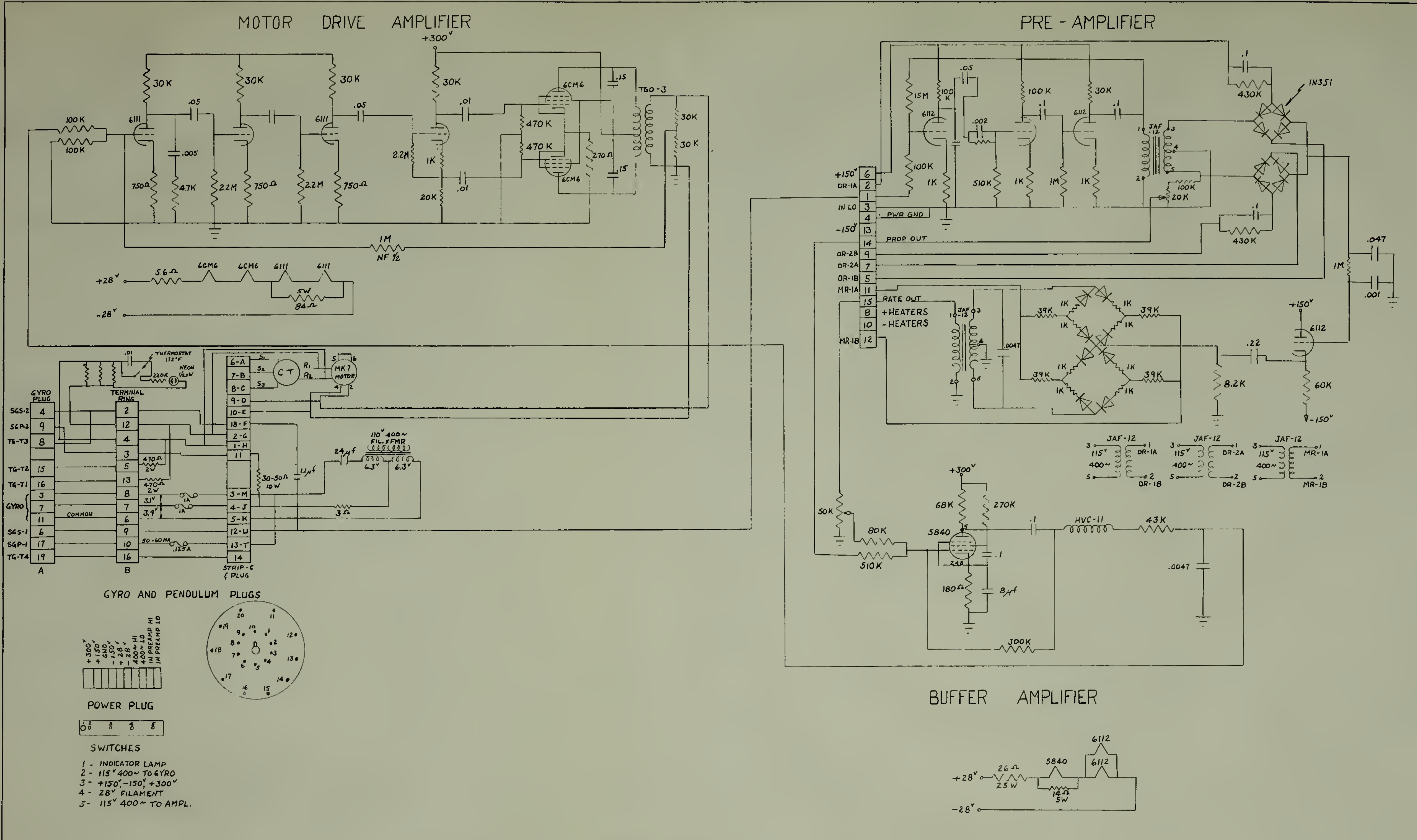
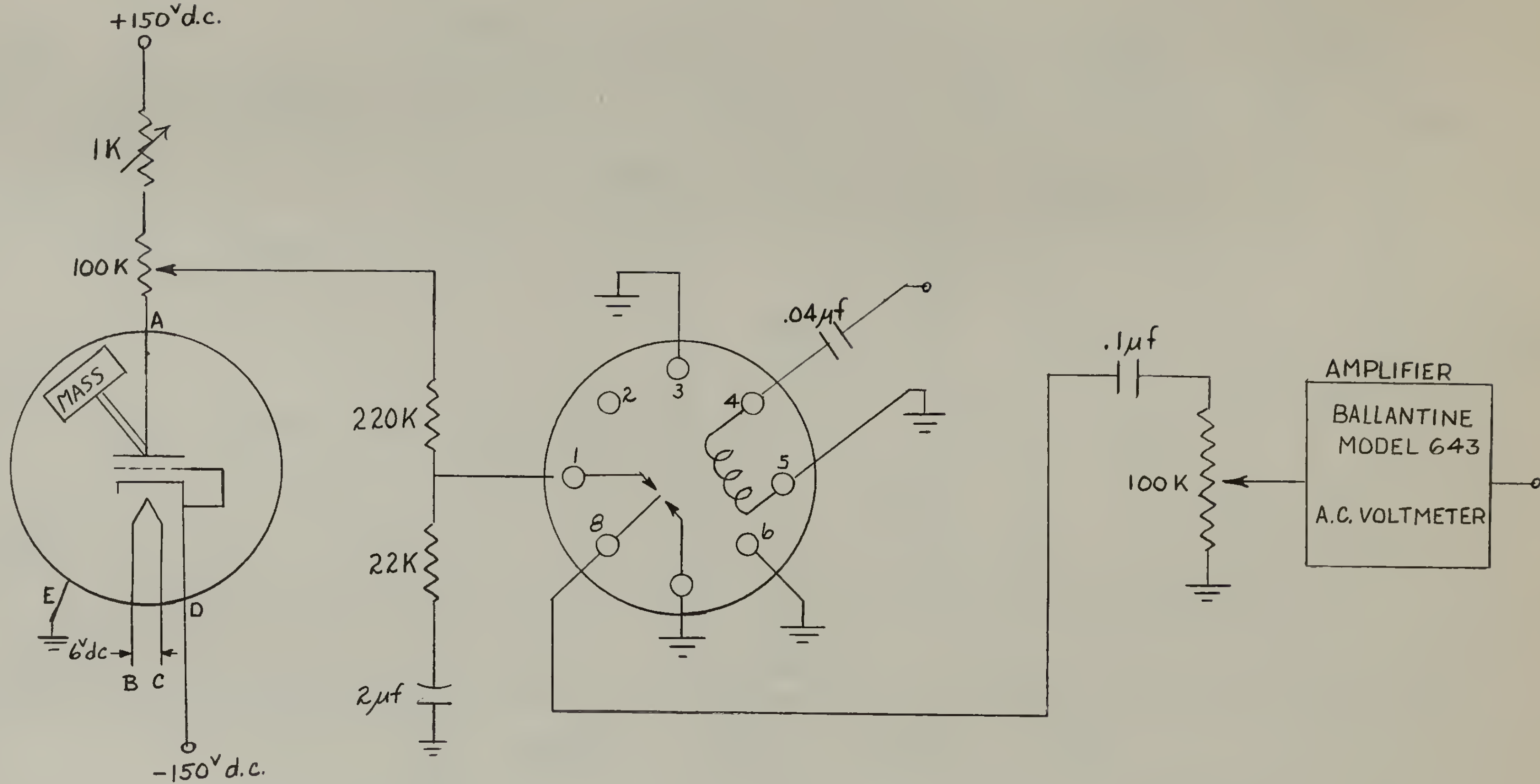


Figure 11. Complete Analog System on Laboratory Bench.



HIG - 4 STABILIZED PLATFORM CONTROL CIRCUIT

FIGURE 12



ACCELEROMETER
CALIDYNE MODEL 18D-5

CHOPPER
AIRPAX MODEL A-580

FIGURE 13. ACCELEROMETER AND CHOPPER CIRCUIT DIAGRAM

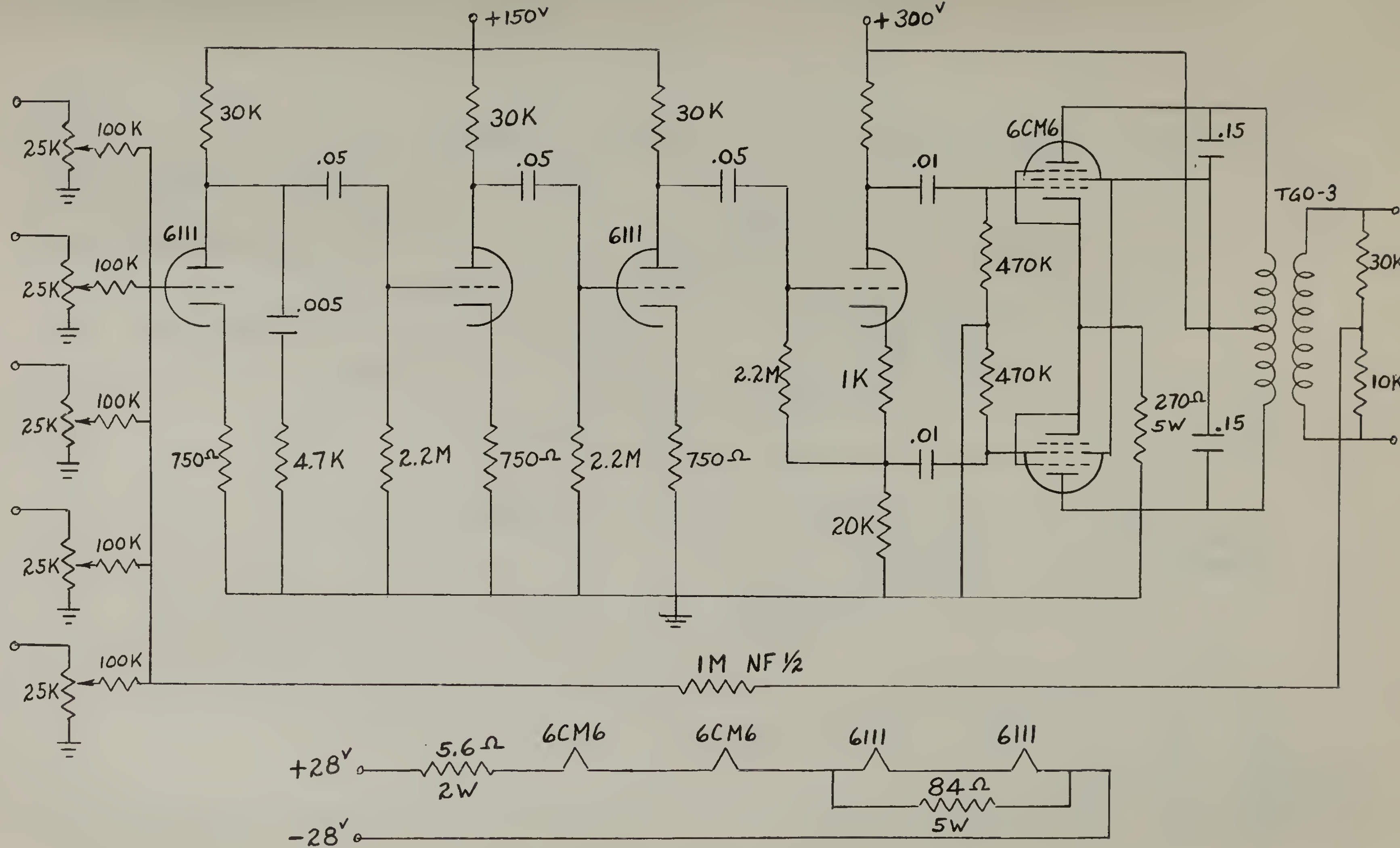


FIGURE 14. FIN DRIVE MOTOR AMPLIFIER

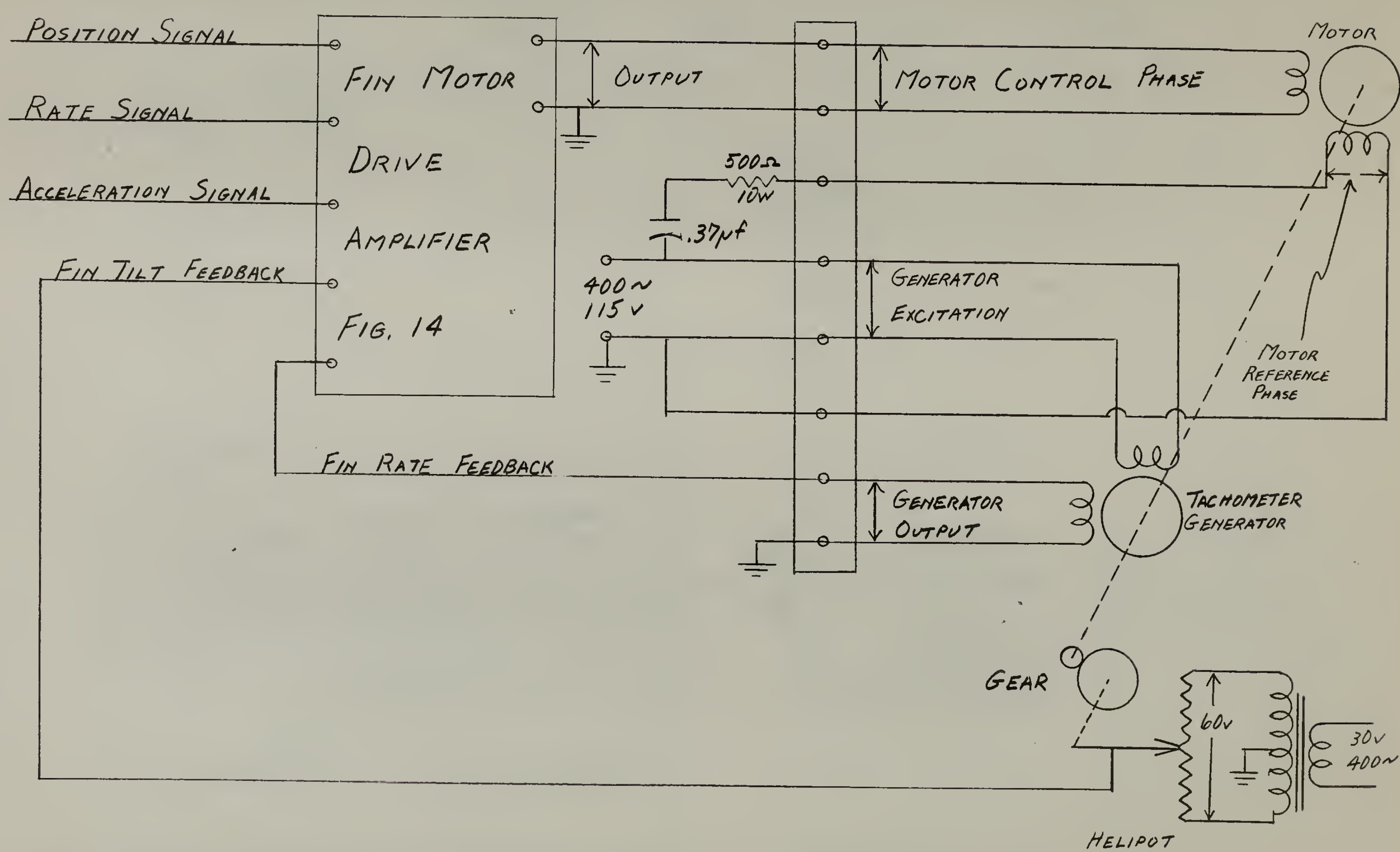


FIGURE 15. SCHEMATIC OF FIN CONTROL SERVO

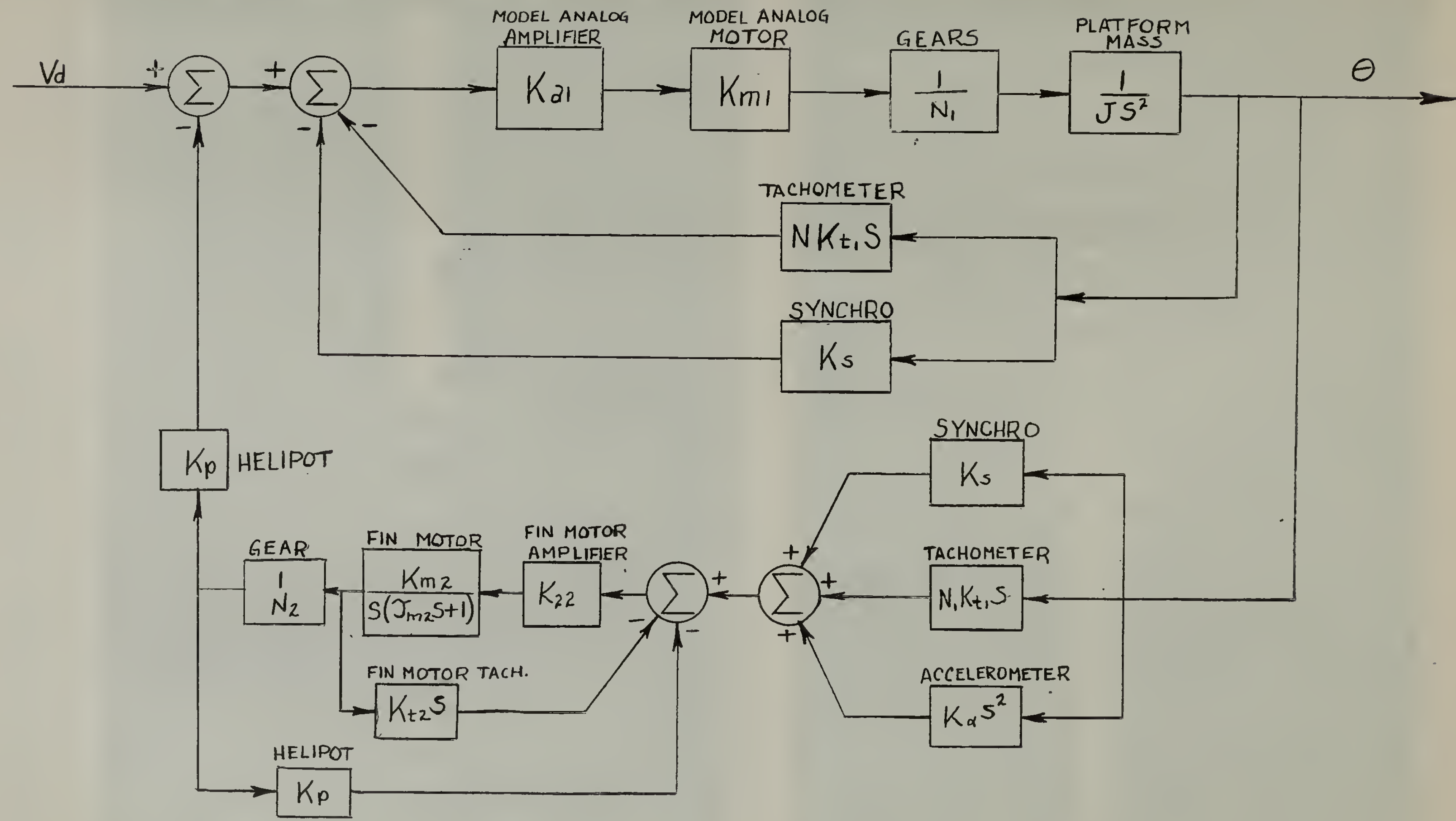


FIGURE 16. BLOCK DIAGRAM OF ENTIRE ANALOG SYSTEM

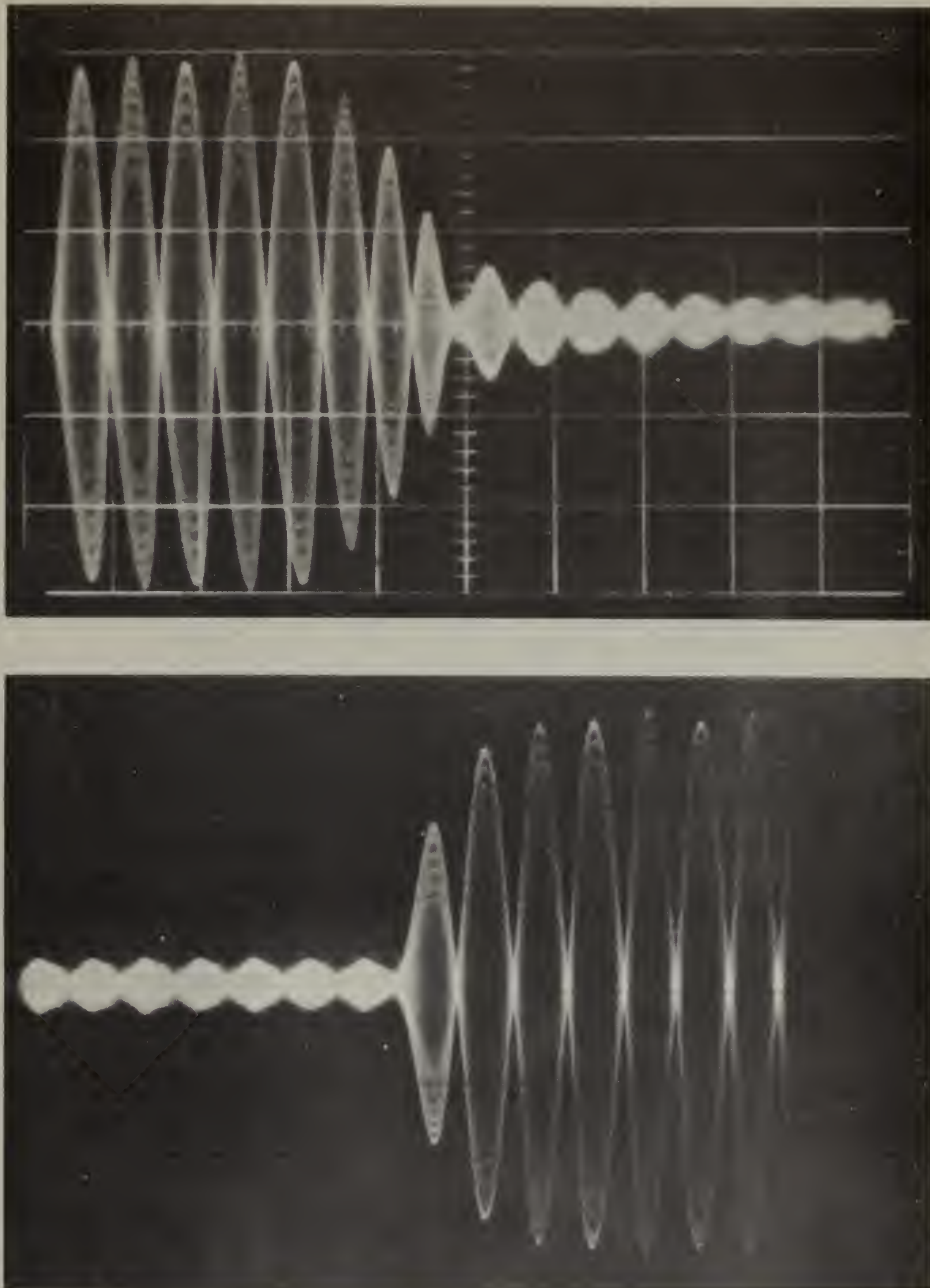


Figure 17. Oscilloscope Photographs of Stable Platform Synchro Voltage. Amplitude is Proportional to Roll Angle of Analog. Upper Photograph is during Stabilization, when Fin Torque Feedback Loop is Closed. Lower Photograph Shows Opening of Fin Torque Loop.

REFERENCES

1. Chadwick, J.H., Jr., "On the Stabilization of Roll", Paper presented before the Society of Naval Architects and Marine Engineers, New York, Nov. 9, 1955.

2. Morris, A.J. and Chadwick, J.H., "Ship Stabilizer Cost in Weight, Space, and Power as a Function of System Parameters", Technical Report No. 3, under contract N6-ONR-25129 for ONR, Stanford University, Stanford, California, Feb. 1953.

3. Allan, J.F., "The Stabilization of Ships by Activated Fins", INA, Vol. 87, 1945, pp. 123-149.

4. Morris, A.J., and Chadwick, J.H., "The Anti-Roll Stabilization of Ships by Means of Activated Tanks", Part B, "Verification of Basic Theory by Model System Studies", Technical Report No. 15 under Contract N6-ONR-25129, for OAR and ONR, Stanford University, Stanford, California, January 1951.

5. Chadwick, J.H., Jr., "The Anti-Roll Stabilization of Ships by Means of Activated Fins", Technical Report No. 1, Part B, "Controls and Performance for Stabilizing Fins", under Contract N6-ONR-25129 for ONR, Stanford University, Stanford, California, February 1953.

6. Rossell, H.E., and Chapman, L.B., "Principles of Naval Architecture", Vol. II, Society of Naval Architects and Marine Engineers, New York, 1941.

7. Labberton, J.M., "Marine Engineer's Handbook", McGraw-Hill Book Co., Inc., New York, 1945.

8. Porter, W.R., "Angular Motion Instrumentation for Model Seaworthiness Testing", Tech. Memo. 6897-TM-10, Contract NOrd 11799, M.I.T. Servomechanisms Laboratory, Cambridge, Mass., May 1955.

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K587 Klingensmith
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